Paper D2-3

SYMPOSIUM ON PUMP DESIGN, TESTING AND OPERATION

OBSERVATIONS ON THE PERFORMANCE OF CENTRIFUGAL PUMPS AT LOW REYNOLDS NUMBERS

by

A. J. Acosta and A. Hollander

To be presented at a Symposium on Pump Design, Testing and Operation, which will be held at the National Engineering Laboratory, East Kilbride, Glasgow, from 12th-14th April, 1965. The papers and discussions are to be published by H.M. Stationery Office in the Proceedings of the Symposium.

BIOGRAPHICAL NOTE

A. J. Acosta graduated B.S. from the California Institute of Technology in 1945 and was awarded the degree of Ph.D. in 1952. He then joined the staff of the Hydrodynamics Laboratory of that Institute and now has the status of Associate Professor of Mechanical Engineering.

A. Hollander graduated M.E. from the Joseph Royal University, Budapest, in 1904. He joined the staff of the Hydrodynamics Laboratory of the California Institute of Technology in 1944 and was appointed Professor of Mechanical Engineering. In 1951 he was granted the status of Professor Emeritus.

OBSERVATIONS ON THE PERFORMANCE OF CENTRIFUGAL PUMPS AT LOW REYNOLDS NUMBERS

by

A. J. Acosta, B.S., Ph.D. and A. Hollander, M.E.
(California Institute of Technology, Pasadena, USA)

SUMMARY

From the results presented herein it is clear that the viscous flow in the centrifugal pump impeller is very complicated, but we see that the internal flow itself is not necessarily too inefficient and further that the hydraulic torque does not necessarily increase with lower Reynolds numbers. These results mean that the effect of external disc friction and its interaction in the volute are as important as the impeller design itself for such viscous flows. The present tests suggest that a pump which works well at high Reynolds numbers, as for water application, will not necessarily be the best pump for the low Reynolds number application. These internal flows are certainly very complicated and the authors hope that some of the techniques and suggestions made herein will be used for such work by others in the future.

Notation

A Area

g Gravitational constant

H Head

p Pressure

Q Flowrate

R Radius

Re Reynolds number ($R^2\ \omega/\nu$ for the impeller)

T Torque

u Impeller tip speed

v Absolute velocity

α Direction of absolute velocity relative to periphery

ν Kinematic viscosity

ρ Density

ψ Head coefficient gH/u_2^2

ϕ Flow rate coefficient (Q/A_2u_2)

τ Torque coefficient $(T/\frac{\rho}{2} A_2u_2^2)$

η Efficiency $(\eta = \phi\psi/\tau)$

ω Angular speed (rad/s)

Subscripts

1 Inlet

2 Discharge

1. Introduction

A number of years ago the California Institute of Technology and the Byron Jackson Company of Los Angeles engaged in a cooperative programme to investigate some aspects of the flow in pumps at Reynolds numbers down to about 2000 based on tip speed and radius. These studies were prompted by a requirement to pump very viscous crude oils. The typical viscosities of the contemplated oils ranged up to about 7000 Saybolt seconds universal (S.S.U.) For the types of pump contemplated, the corresponding Reynolds numbers ranged from about 2000 to 6000. It was known that the performance of centrifugal pumps at these low values of Reynolds numbers could be expected to be, and was, very poor indeed. For example, Ippen(1) indicates that the expected efficiencies for a Reynolds number of 5000 would be on the order of 30 per cent, which is very low. At the same time it was realized, and it was brought out by Ippen's work, that a major factor in the decreased efficiency was the influence of disc friction. However, it was not known whether the internal fluid flow experienced some sudden or drastic change to help account for the very low efficiency observed at these low Reynolds numbers.

To explore these factors a dual programme was, as already mentioned, arranged between the Byron Jackson Company and the Hydrodynamics Laboratory at the California Institute of Technology. The tests conducted by the Byron Jackson Company were to be carried out on complete pumps with the emphasis on modifications of impeller design for operation at low Reynolds number and to attempt to reduce the disc friction on the external shroud surfaces of the rotating impeller. At Caltech, on the other hand, the emphasis was on examination of the internal flow in the impeller and on flow visualization studies to find what were the important features of the flows at these low Reynolds numbers which might account for a possible low efficiency.

The experiments to be described herein were intended to be, not on complete pumps, but rather to observe and obtain qualitative information on the internal flow within a centrifugal impeller over a fairly wide range of Reynolds numbers. Thus, disc friction and volute losses were not subjects of direct concern.

2. Experimental Approach

For the purposes of the present experiment, it was desirable to provide experimental equipment which would make it possible to carry out flow visualization within the impeller, and at the same time to measure the normal pump characteristics, e.g., head, flow rate, torque, efficiency, etc. It was also desired to measure internal flow distributions of velocity, total pressure loss, etc. At the same time the Reynolds number range was rather large and it was realized that to avoid a complex piping circuit, mixing tanks and the like, a simple flow circuit was very desirable for reasons of both cost and flexibility. The working fluid selected was glycerin. The rotating speeds of the impellers ranged from about 200 to 500 rev/min. The geometrical parameters of the model impellers used are given in Table 1. The discharge diameter was about one foot and with this combination of size and speed a Reynolds number range from about 1500 to 600 000 could be easily achieved. The variation of Reynolds number was accomplished by varying both the rotational speed and the viscosity of the working fluid. The fluid viscosity was varied by diluting the glycerine with water. Glycerin is an excellent fluid for tests of this sort as it is perfectly Newtonian, stable, non-toxic and non-flammable. One slight disadvantage of dilute glycerin, however, is that the viscosity is greatly dependent upon temperature and in certain concentration ranges is a very sensitive function of the amount of diluent. To reduce the amount of working fluid required, a self-contained recirculating loop was designed and a schematic cross-section is shown in Fig. 1. It can be seen here that the flow recirculates through the impeller, through a collecting vaneless diffuser and thence back into the impeller inlet in a sort of toroidal flow path. Throttling arrangements were provided both at the bell inlet at the bottom of the tank and at the exit of the vaneless diffuser. The original intention was to adjust the position of the two throttle valves so that the pressure at the exit of the impeller would be slightly greater than atmospheric. It was also intended to run dry the lower, as well as the upper, surface of the impeller so that a minimum amount of disc friction would be present. The impeller was sealed on its suction shroud with a carbon facing ring which was lightly held against the impeller shroud by means of a slightly inflated tube.

This arrangement minimizes the amount of working fluid and the flow paths are at all times symmetrical with respect to the impeller. At the same time the velocity distribution approaching the impeller inlet is as good as can be arranged at these low Reynolds numbers. Certain measurement problems do arise, however. Of these the one posing the greatest difficulty is the measurement of flowrate. For this purpose the pressure drop across the contracting nozzle immediately upstream of the impeller was calibrated as a function of flowrate for several Reynolds numbers. This was made possible by measuring the velocity distribution at the throat section of the nozzle by means of a remotely actuated impact probe. In carrying out this calibration it was necessary to account for the known real-fluid effects on the impact probe.

Torque was measured in a conventional way by reacting the torque on the suspended dynamometer case against known weights. Total head was measured by taking the difference in the reading of a total head probe $\frac{5}{8}$ inch beyond the discharge of the impeller and another total head probe on the centreline of the inlet nozzle.

3. Operating Procedure

It was found that it was not possible to maintain easily a close running clearance between the rotating impeller and the stationary vaneless diffuser and still keep the lower surface of the impeller dry. As a result the lower shroud of the impeller was allowed to be flooded and the disc friction due to this and due to the running clearance between the impeller and vaneless diffuser was carefully measured and subtracted from the observed torque. In this way only the torque required by the internal flow through the impeller was found.

Determining the viscosity of the fluid, especially when the glycerin was nearly pure, presented somewhat of a problem. The fluid temperature varied slightly from day to day, but sufficiently to make a significant difference in the viscosity. As a result it was not possible to carry out the measurement of the impeller performance characteristics at exactly the same Reynolds number in successive tests. The viscosity of the glycerin-water mixture was continuously monitored with a Brookfield viscometer. The pressure drop across the nozzle, and total head, were measured with Statham differential pressure transducers. These were calibrated with water manometers. The experiments were then fairly straightforward, although the calibration of the inlet contracting nozzle was rather tedious.

4. Experimental Results

To give a qualitative idea of the flow distributions downstream of the impeller, a few velocity profiles were measured at the discharge plane of the impeller $\frac{5}{8}$ inch beyond the impeller tip. Fig. 2 shows the distribution of absolute leaving angle across the breadth of the impeller. It can be seen that there is a region of back flow at this station for the upper 15 per cent of the diffuser. This observation, though made in water at the highest Reynolds number (6×10^5), was typical at lower Reynolds numbers also, and it represents the kind of flow separation that can be expected in diffusers of this type. Absolute velocity profiles across the passage are shown in Fig. 3 for various Reynolds numbers. This shows that at the higher flow coefficients the boundary layer on the impeller shrouds has not penetrated to the centre of the passage. At the lowest flowrate (and at the lowest Reynolds number also) a more-or-less parabolic profile of absolute velocity is observed. Although these measurements are made at the $\frac{5}{8}$ inch station beyond the 23.5 degree impeller, they are typical of the 45^o and 55^o impellers also. Of some incidental interest is the pressure recovery within the vaneless diffuser itself. This is shown in Fig. 4 for several Reynolds numbers and flow coefficients. The pressure recovery is mainly affected by the flow coefficient and not by the Reynolds number.

Of primary concern is the distribution velocity and flow angles within the impeller. As it turns out, time did not permit us to measure internal velocity distributions or losses even though provisions had been made for this purpose. However, extensive tuft studies were carried out in all the three impellers. Some idea of the large effect that viscosity has on the internal flow patterns can be obtained from Fig. 9, where relative flow angles near the exit of the impeller were determined by the tuft studies. It can be seen that a major feature at these low Reynolds numbers is the secondary flow that develops within the passages of the impeller. Fig. 5 shows that in the centre of the passage the boundary layer is overturned by 20^o to 25^o. As this boundary layer is thick, this represents an appreciable change in relative flow angles and can be expected to have a major effect on the torque required of the impeller. The photographs of the tufts from which the data of Fig. 5 were taken are shown in Fig. 6. In both photographs the Reynolds number is 2200 and the flow coefficient is near that corresponding to the best efficiency for the 23.5° impeller when operating in water. In both photographs the boundary layer overturning on both shrouds, top and bottom, can be easily seen. No drastic zone of separation seems to be evident and the tufts in the middle of the channel breadth are well behaved.

We will now summarize the measurements made of the overall impeller characteristics. It should be remembered that these curves refer only to the internal flow through the impeller and that no allowance has been made for disc friction. The total head was measured in two ways. As mentioned before, one of the ways was to measure the discharge total head with an impact probe at a distance $\frac{5}{8}$ inch beyond the impeller discharge. In the interest of simplicity and in making comparative measurements, the total head probe was installed in the middle of the diffuser passage and was always adjusted to give the maximum reading. Of course this does not necessarily represent the correct average total head. In fact, it is almost impossible to get a correct average total head when strong non-uniformity in the flow exists from one vane to the next. Wakes, etc., in the relative flow will, for example, give rise to disproportionately high total pressure. The second method was to measure the static pressure in the vaneless diffuser at a radius of 2.2 times the radius of the impeller. This latter value may be more representative of actual pump performance as it offers or incorporates some measure of diffuser loss and mixing loss that should be charged to the impeller. The results of the latter measurements on the three different impellers are shown in Fig. 7.

From these figures it is quickly apparent that the effect of viscosity is different for the three types of impellers. The performance of the 23.5° impeller (shown in Fig. 7(a)) indicates that the torque required continuously, increases as the viscosity increases or Reynolds number decreases. The total pressure rise (measured with the static pressure in the diffuser as mentioned) is somewhat below the water performance ($Re = 6 \times 10^5$) and is about 15 per cent less than the total pressure measured with the impact probe at the impeller discharge. The efficiencies, however, are not terribly low. For experiments at the Reynolds number of 7000 we observe an efficiency of about 65 per cent. The other two impellers, and especially the 55° impeller, show quite a different behaviour with Reynolds numbers. Typically, of course, the total pressure rise is always less at a higher viscosity than at a lower viscosity. But what is most surprising is the behaviour of the required torque with Reynolds numbers. It can be seen on both (b) and (c) of Fig. 7 that the maximum torque requirement occurs at the highest Reynolds number and that when the Reynolds number decreases the maximum torque decreases. Furthermore, the flowrate for the maximum torque also decreases with decreasing Reynolds numbers. Again at a Reynolds number of between 8000 to 10 000 the measured efficiency amounts to about 62 per cent. Fig. 8 also show similar results (except that the torque is not shown) for the total pressure measurements made at the impeller discharge. From these figures it can be seen that for the lower Reynolds number the higher vane angle performs relatively better than the lower vane angle. That is to say, a pump designed to work for the water (i.e., high Reynolds number) does not perform as well as a pump with larger blade angles which normally would not be a satisfactory water pump.

Incidentally, these measurements can lay no great claim to accuracy. The reasons for this are several. First of all, an instability at the lower Reynolds numbers occurs at a flow coefficient of about 0.1. A strong unsteady irregular pulsation then develops at the exit portion of the impeller. The absolute flow may then vary by as much as 25 per cent in torque. Tufts placed within the impeller passages may even show occasional reverses. The origin of this pulsation is probably within the vaneless collector and not the impeller although this point is at present obscure. Such pulsation also occurs at the higher Reynolds number (6×10^5) although not until the flowrate coefficient is much lower - e.g., 0.05 or so. An additional source of error may also lie in the calibration of the inlet contracting nozzle. This is not expected to be large, but this possibility certainly exists. All of these indicate that the error in the efficiency determination may amount to about 3 per cent. This is brought out in Fig. 8(a), which shows that the efficiency of the water tests for the 23.5° impeller is just about 100 per cent. From previous work on two-dimensional impellers, we know that this is too high by about 3 or 4 per cent at these Reynolds numbers. Incidentally, these measurements are taken at the mid-passage and do not represent a proper average of the total head across the passage. Another difficulty of these experiments was that it was not possible to measure the torque, head, and flowrate simultaneously. Consequently the Reynolds number for the torque determination did not always agree precisely with the Reynolds number for the total head determination, etc. As a result these curves can only be regarded as trends and the Reynolds numbers are really bands. This accounts probably for the 'wavy' appearance of some of these head characteristics.

Extensive internal flow visualization studies were carried out upon all impellers. A sequence of photographs showing tufts on the surfaces of the vanes and shroud surfaces of the 45° impeller are shown in Fig. 9. There are a number of interesting features in these photographs. First, all of the tufts show clearly that even at this low Reynolds number (1700) the flow is attached to the suction shroud surface at all times - even with the fairly abrupt turn on this surface. Evidently the stabilizing effects of the rotation of the shroud boundary enable this very viscous flow to turn the corner. Also apparent in all photographs is the very large amount of overturning of the boundary layer. We can contrast, for example, the tuft figures shown in C, D and E which are on the bottom shroud of the impeller with tufts shown in the middle of the impeller passage in G. Also interesting is that even at low flow coefficients, as shown in J, no gross separation on the suction surface of the vane is evident. Principally, what happens is that a very thick boundary layer develops which gives rise to the secondary flow visible in these photographs. At the lower flow coefficients, especially J and K, it is clear that a strong flow reversal on the pressure surface of the vane is taking place. This is, however, well outside the normal operating portion of the impeller.

Similar results were observed on the 55° impeller. Incidentally, the leading edges of the vanes shown in Fig. 9 are semi-circular. Additional tuft studies were carried out with the vanes sharpened like a hatchet blade. If anything, the flow appeared better behaved with the sharpened blades than with the round ones, and there are even indications that the efficiency was better with sharpened blades.

An additional qualitative remark may be made here. A thread streamer was added to the probe in the inlet of the nozzle before the impeller. Since the impeller drop shroud is transparent, it was possible to determine whether or not a significant pre-whirl was present in the inlet flow. It was found that for all Reynolds numbers the flow into the impeller inlet was purely axial for flowrate coefficients greater than 0.02, except for a very thin layer (approximately $\frac{1}{16}$ inch) near the rotating portion of the inlet shroud. For flowrates smaller than this a strong back flow and ring vortex develops. However, this phenomenon is of little interest for the normal portion of the pump characteristic.

5. Further Remarks

The trends of the present tests were generally confirmed by tests on complete pumps carried out by the Byron Jackson Company. From both sets of experiments it appears that the major source of poor efficiency at low Reynolds numbers is the disc friction on the shroud and the loss in the volute. The Byron Jackson Company results confirm the general behaviour of the impeller results discussed in this paper. They made a number of attempts to reduce the disc friction. Of great interest among these various attempts was the adoption of a scheme first proposed by Ňovák in 1905[2]. He used a very low specific speed, inefficient impeller, followed by a radial vaneless diffuser which led into a constant section volute with a discharge nozzle. The vaneless diffuser is mounted on two freely rotating shrouds which will turn between the impeller shrouds and the stationary housing, thus reducing the disc friction and, improving the flow through the diffuser, by turning it. It is hoped that this forgotten scheme will be re-examined and its possibilities fully explored.

REFERENCES

1. IPPEN, A. T. The influence of viscosity on centrifugal-pump performance. *Trans Amer. Soc. Mech. Engrs*, 1946, **68**(8), 823-848.

2. NOVAK, J. Even diffusers on centrifugal pumps (in German). *Z. ges. Turbinewesen*, 1907, **4**.

TABLE 1

Impeller Constants

Impeller Designation		55	45	23.5
Inlet angle	(degrees)	26	23.5	23.5
Discharge angle	(degrees)	55	45	23.5
Discharge diameter	(in)	10.3	10.3	10.3
Radius ratio		0.59	0.59	0.59
Breadth	(in)	1.20	1.20	1.20
No of vanes		10	6	6
Vane thickness	(in)	0.1	0.1	0.1

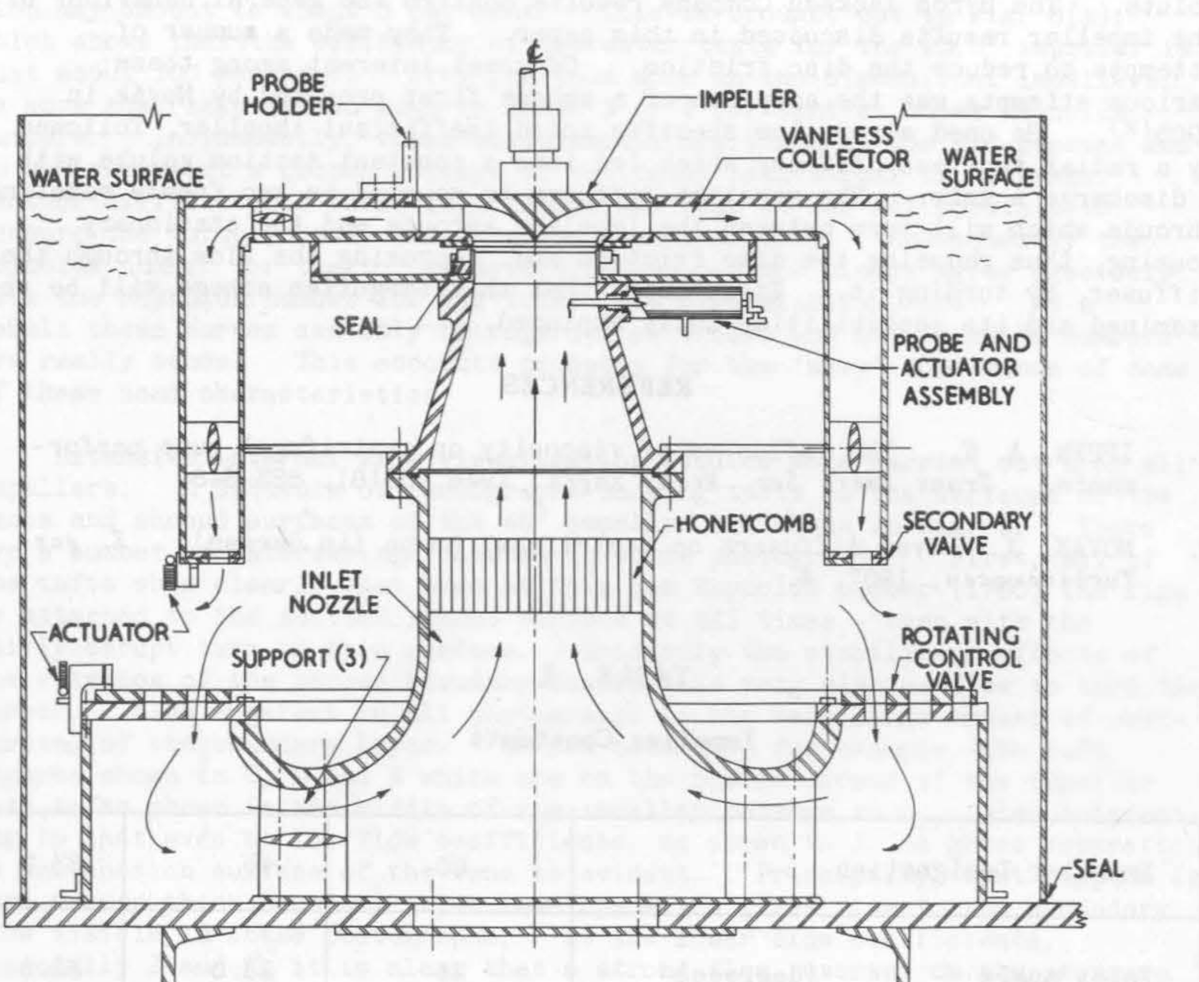


FIG I SCHEMATIC OF CIRCUIT

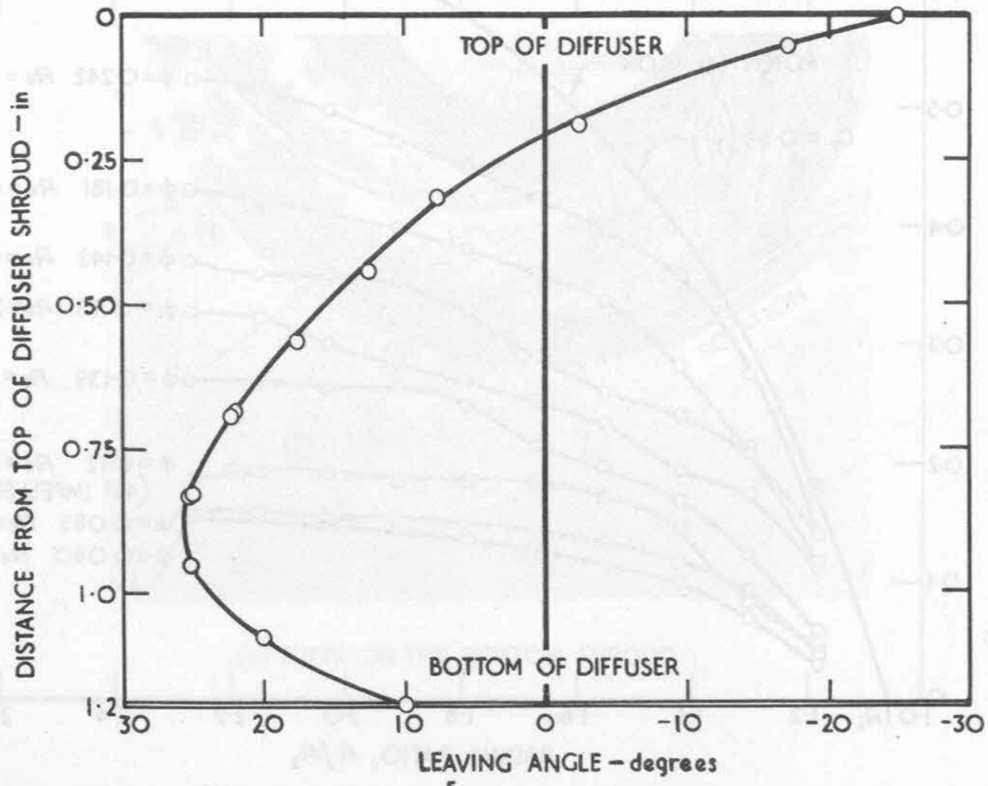


FIG 2 ABSOLUTE LEAVING ANGLE $\frac{5}{8}$ in FROM DISCHARGE OF IMPELLER. IMPELLER DISCHARGE ANGLE=23·5°, FLOW COEFFICIENT=0·12, $Re=6\times10^5$

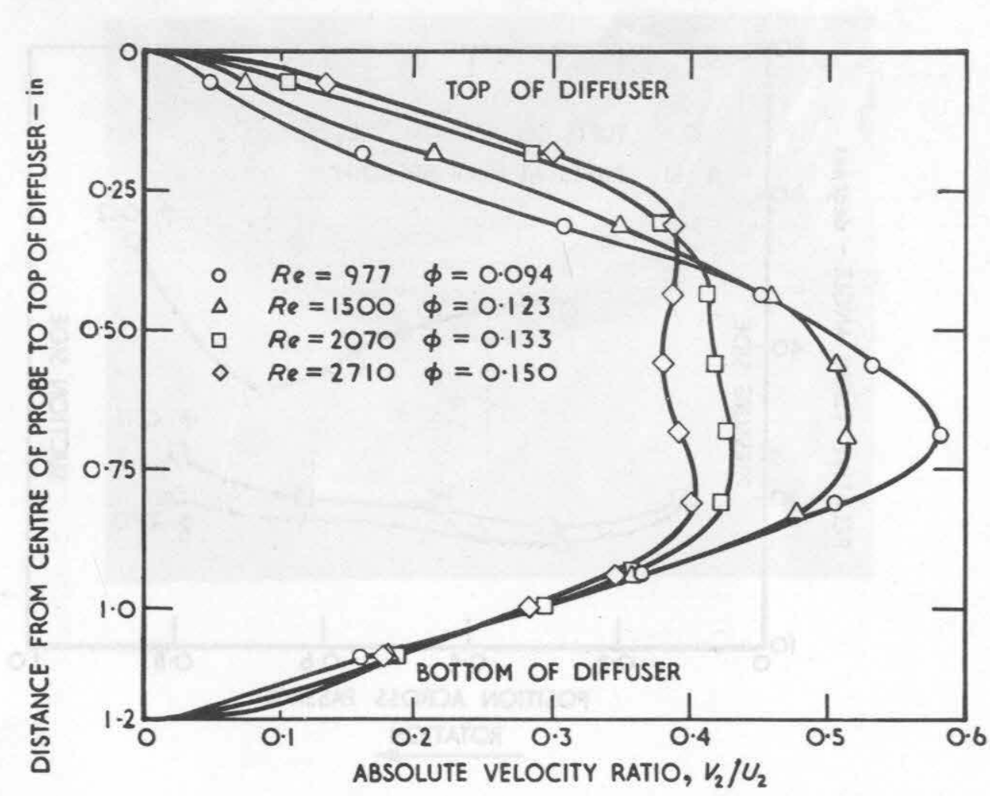


FIG 3 ABSOLUTE VELOCITY DISTRIBUTION $\frac{5}{8}$ in FROM DISCHARGE OF 23·5° IMPELLER

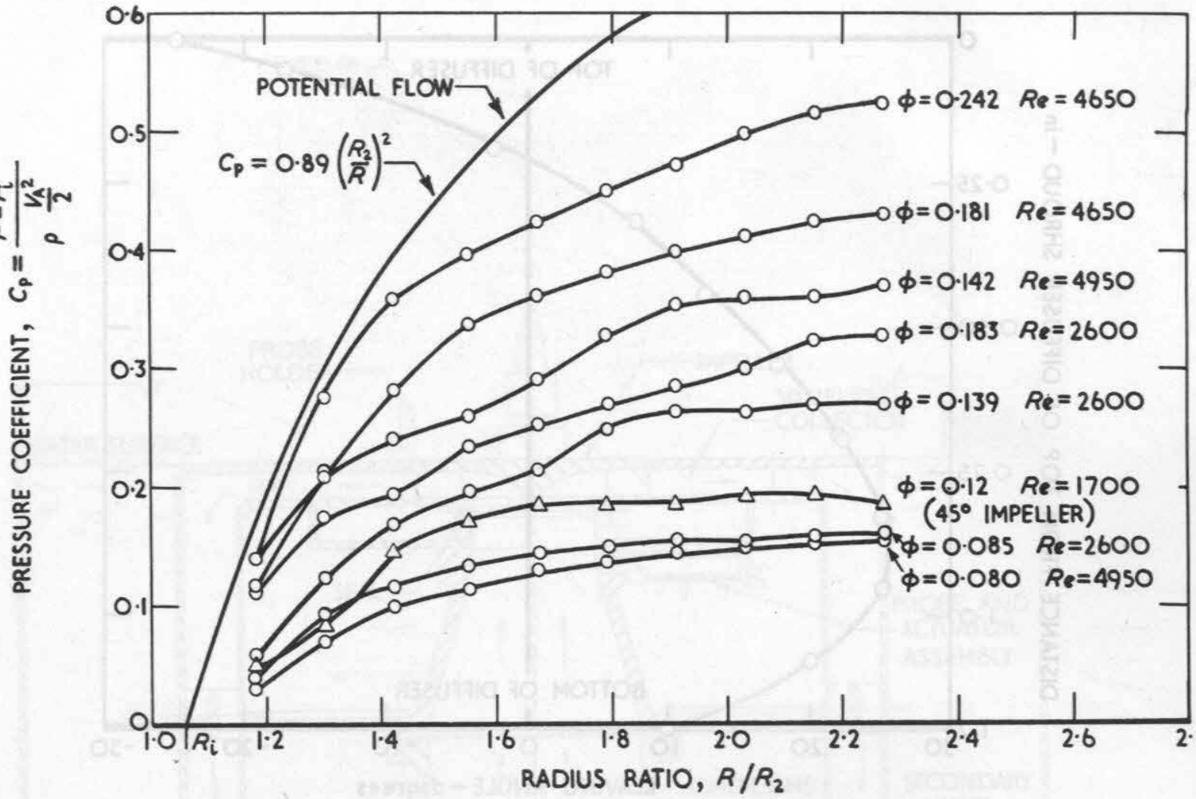


FIG 4 STATIC PRESSURE RECOVERY IN VANELESS DIFFUSER. MOST OF THE POINTS ARE FOR THE 55° IMPELLER, EXCEPT ONE WHICH IS FOR THE 45° IMPELLER

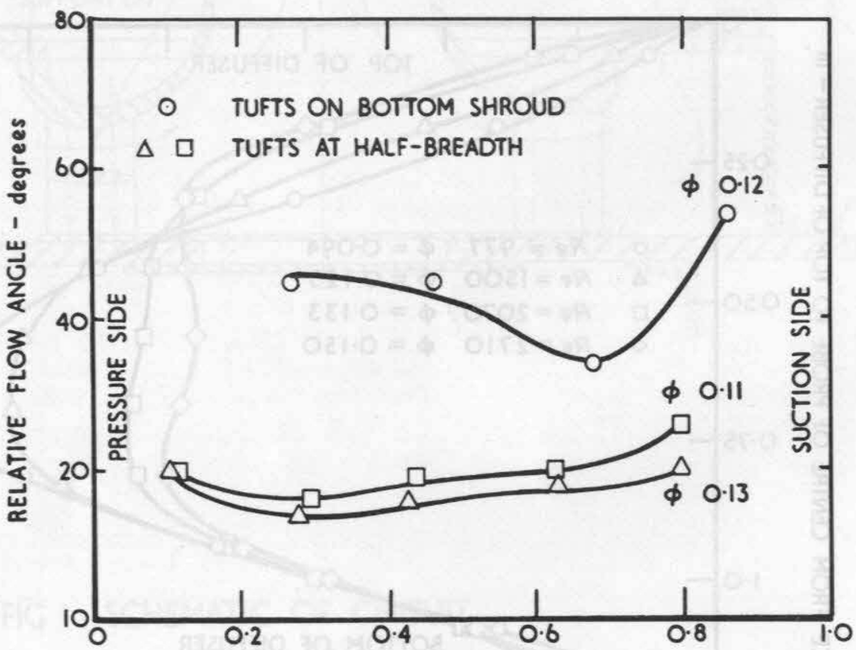
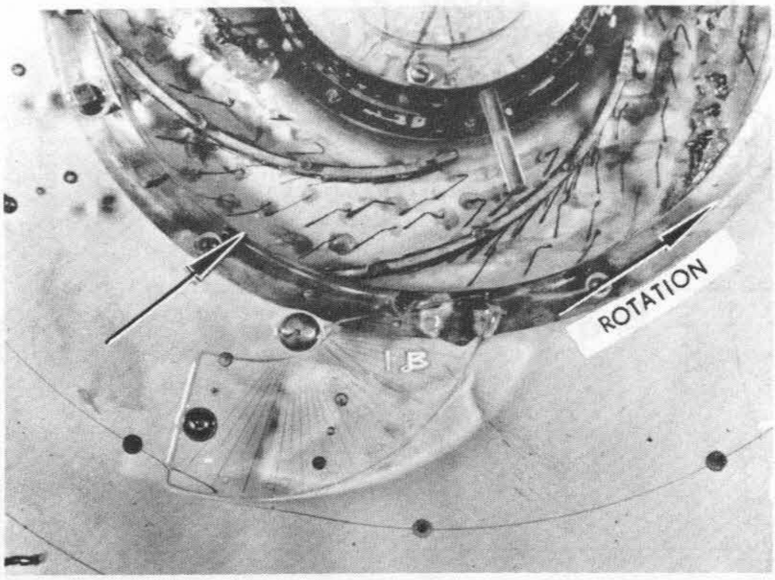


FIG 5 RELATIVE FLOW ANGLES ON THE BOTTOM SHROUD AND IN THE MIDDLE OF THE IMPELLER FOR THE 23·5° IMPELLER AT $Re = 2200$



(a) TUFTS ON THE BOTTOM SHROUD

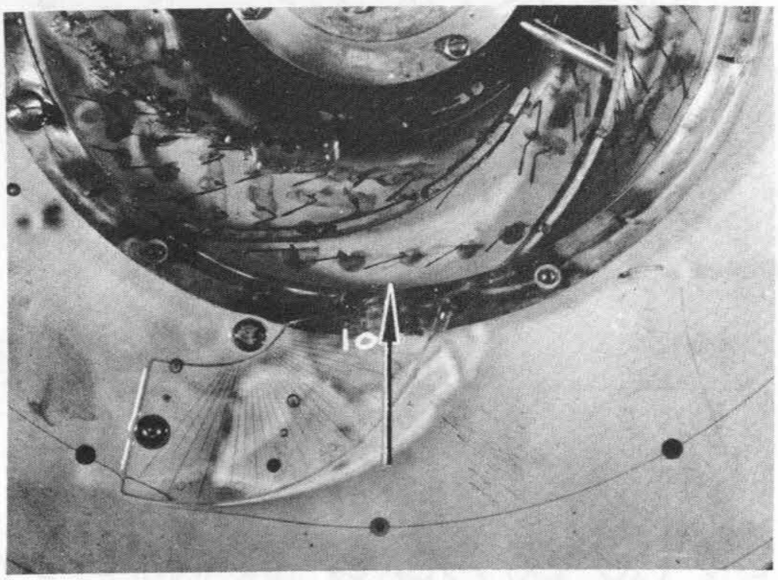
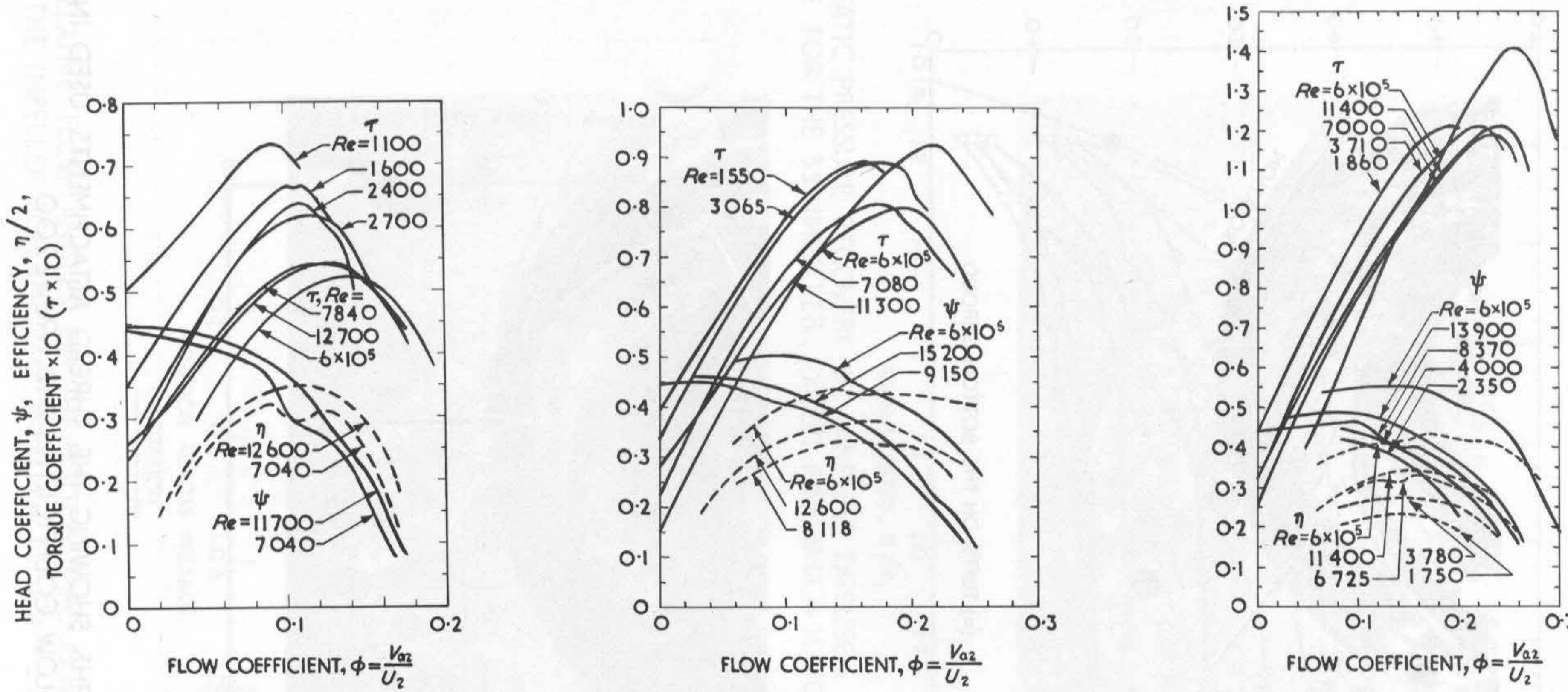


FIG 6 PHOTOGRAPHS SHOWING THE THREAD ATTACHMENTS USED IN FIG 5. FLOW COEFFICIENT= 0·12, $Re = 2200$



(a) 23·5° IMPELLER (b) 45° IMPELLER (c) 55° IMPELLER

FIG 7 HEAD COEFFICIENT AND EFFICIENCY v FLOWRATE COEFFICIENT (USING STATIC PRESSURE AT RADIUS OF 2·2 TIMES IMPELLER RADIUS)

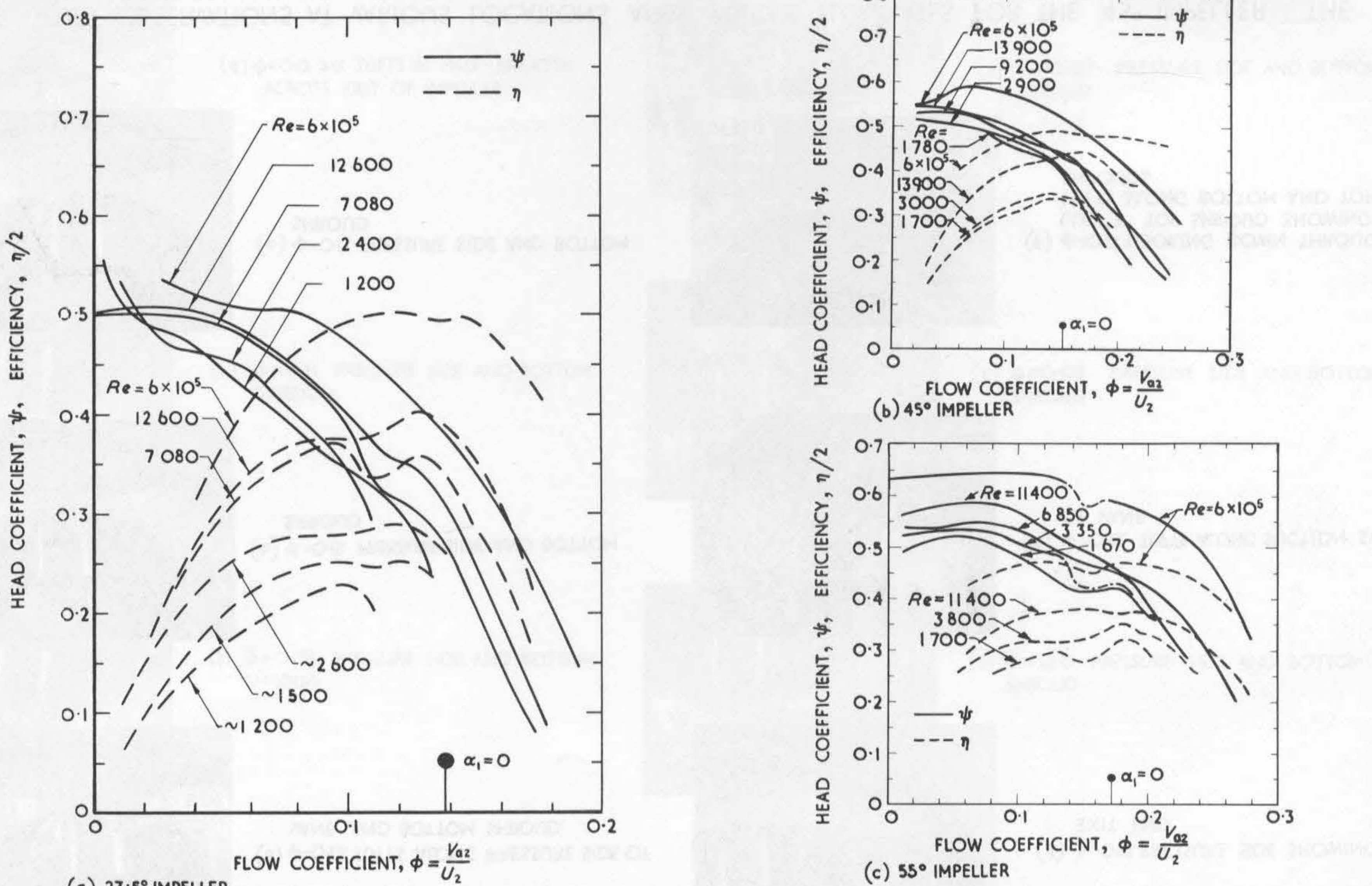
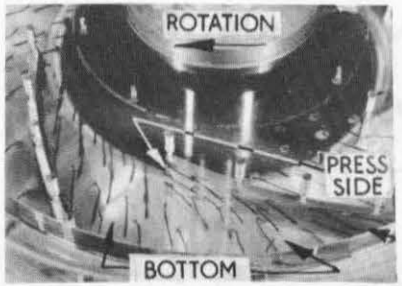
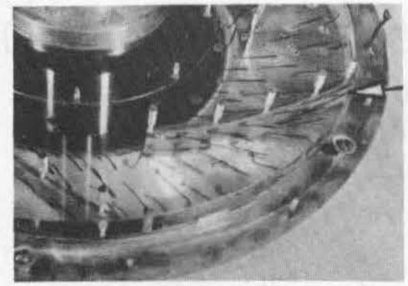


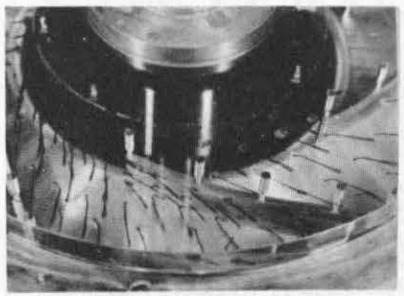
FIG 8 HEAD COEFFICIENT AND EFFICIENCY v FLOWRATE COEFFICIENT (USING YAW INSENSITIVE IMPACT PROBE $\frac{5}{8}$in BEYOND IMPELLER DISCHARGE)



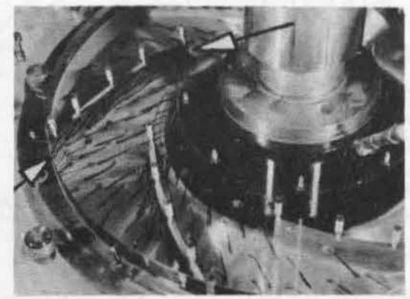
(a) $\phi = 0{\cdot}17$. TUFTS ALONG PRESSURE SIDE OF VANE AND BOTTOM SHROUD



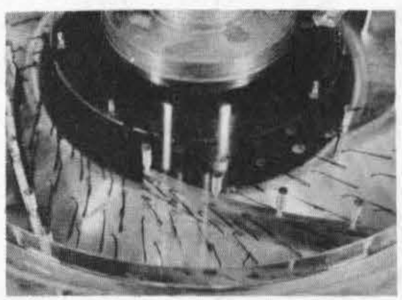
(d) $\phi = 0{\cdot}13$. PRESSURE SIDE SHOWING EXIT END



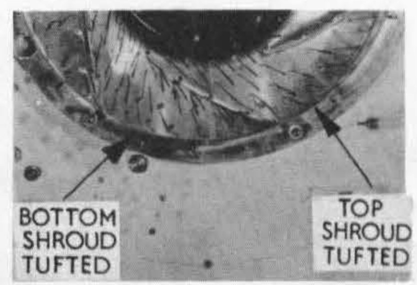
(b) $\phi = 0{\cdot}15$. PRESSURE SIDE AND BOTTOM SHROUD



(e) $\phi = 0{\cdot}13$. TUFTS ALONG SUCTION SIDE OF VANE

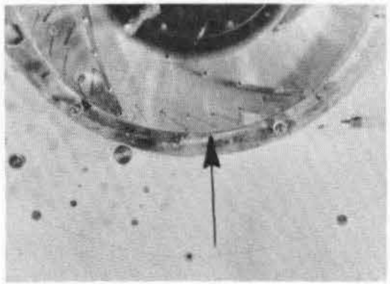


(c) $\phi = 0{\cdot}13$. PRESSURE SIDE AND BOTTOM SHROUD

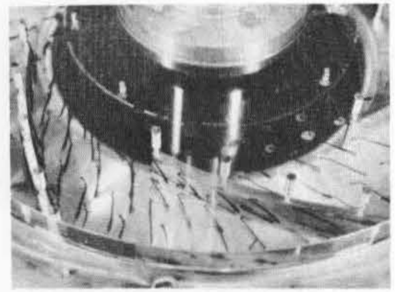


(f) $\phi = 0{\cdot}13$. LOOKING DOWN THROUGH LUCITE TOP SHROUD SHOWING TUFTS ALONG BOTTOM AND TOP SHROUDS

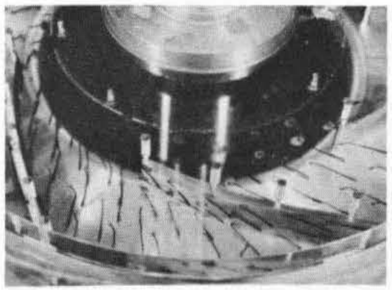
FIG 9 TUFT OBSERVATIONS AT VARIOUS LOCATIONS AND VARIOUS FLOWRATES FOR THE 45° IMPELLER THE ROTATION ON ALL PHOTOGRAPHS IS CLOCKWISE. AVERAGE $Re = 1700$



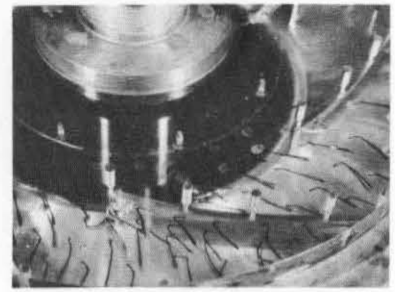
(g) ϕ=0·13 SIX TUFTS AT HALF-BREADTH ACROSS EXIT OF IMPELLER



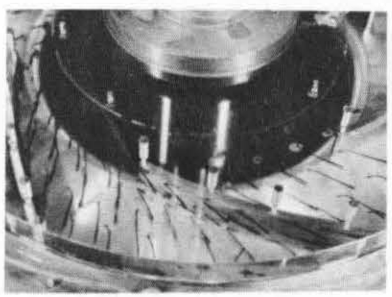
(j) ϕ = 0·07 PRESSURE SIDE AND BOTTOM SHROUD



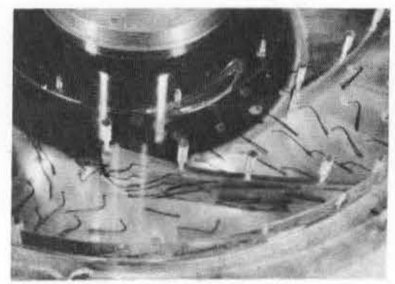
(h) ϕ=0·11 PRESSURE SIDE AND BOTTOM SHROUD



(k) ϕ=0·03 PRESSURE SIDE AND BOTTOM SHROUD



(i) ϕ=0·09 PRESSURE SIDE AND BOTTOM SHROUD



(l) ϕ = 0·0 PRESSURE SIDE AND BOTTOM SHROUD

FIG 9 contd. TUFT OBSERVATIONS AT VARIOUS LOCATIONS AND VARIOUS FLOWRATES FOR THE 45° IMPELLER THE ROTATION ON ALL PHOTOGRAPHS IS CLOCKWISE AVERAGE Re=1700